Fermilab

Fermi National Accelerator Laboratory
Technical Division
Engineering and Fabrication Department
P.O. Box 500
Batavia, IL 60510
Fax: (630) 840-8036

LHC Interaction Region Quadrupole
Engineering Note for Complete Magnet Testing at Fermilab

Rev	Date	Description	Originated by	Approved by
None	December 4, 2000	Original issue	T. Nicol	J. Kerby
A	February 28, 2001	Revised paragraphs 1.0, 2.5.1, 2.5.2, 2.7 based on initial review.	T. Page	T. Nicol
B	March 8, 2001	Revised paragraphs 2.5.1 and 2.5.2 to clarify total combined stress.	T. Page	T. Nicol

LHC Interaction Region Quadrupole
Engineering Note for Complete Magnet Testing at Fermilab

Table of Contents

Chapter 1

LHC Interaction Region Quadrupole

Engineering note for complete magnet testing at Fermilab

1.0 Introduction

This document constitutes the engineering note for the prototype LHC interaction region quadrupoles being fabricated at Fermilab. In addresses the adequacy of the design and installation for testing single magnets at the Magnet Test Facility (MTF) within the Technical Division of Fermilab. Both generic and specific issues are addressed. Generic issues are those which pertain to the individual magnets themselves. Specific issues are those which apply only to the operating modes at Fermilab. For example, relief piping and relief valve analyses and discussions apply only to this specific installation, are not applicable to a string of magnets, and we make no attempt to generalize to that extent.

The magnet, piping and vacuum vessels will not be ASME Boiler and Pressure Vessel Code stamped vessels (hereinafter referred to as "the Code"). We do meet the Fermilab requirement to apply the design rules of the Code such that the intent of the Code is realized, i.e. that the geometry of all welds are consistent with the Code, that allowable stresses are met, etc. Fermilab manufacturing practices do not meet all of the Code requirements, most notably the continuous monitoring of all production processes, radiography of welds, etc. For that reason, Fermilab procedures require that allowable stresses be de-rated to 80% of their Code values. For the design and analysis of internal piping, we have applied the rules and practices outlined in ASME Code for Pressure Piping, B31.3, "Chemical Plant and Petroleum Refinery Piping". We have designed the bellows according to the standards of the Expansion Joint Manufacturers Association, Inc. (EJMA).

The chapters and appendices included in this note address each of the following major magnet systems in detail. Refer to the table of contents for the exact location of each analysis or component.

- Cold mass
- Internal piping
- Vacuum vessel
- Interconnects

1.1 Summary of results

It will be shown in each of the following chapters that the design of each magnet system is consistent with the operating requirements at MTF. Chapter 2 will address the cold mass in detail and will document a maximum allowable working pressure (MAWP) of 175 psi. The system relief settings at MTF are set at or below 100 psi. Chapter 3 will address the design of all internal piping and will show that it satisfies the requirements of ASME B31.3, when subject to the operating temperatures and pressures summarized in table 3.0.1. Chapter 4 documents the design and analysis of the vacuum vessel and shows that it meets the requirements of the Code as it applies to vacuum vessels and to section 5033 of the Fermilab ES&H manual when subject to all the applicable structural loads and the insulating vacuum load. Finally, chapter 5 documents the design and analyses of all interconnect bellows. The requirements, design rules, and calculation guidelines of the Expansion Joint Manufacturers Association (EJMA) were used throughout this chapter. EJMA is the recognized standards organization for bellows and is referred to throughout the ASME Boiler and Pressure Vessel Code.

We believe the designs of the systems documented in this note are adequate to ensure that their operation represents no hazard to personnel or to any of the external systems to which they will be connected.

Chapter 2

LHC Interaction Region Quadrupole

Cold Mass Assembly

2.0 Introduction

The cold mass assembly in an LHC IR Quadrupole consists of the following major components.

- Quadrupole collared coil assembly
- Cold iron yoke
- Outer helium containment vessel

The helium containment vessel consists of the following.

- Two 304L stainless steel skins (half shells)
- Two 304L stainless steel alignment keys
- Two 304L stainless steel end plates
- Two 304 stainless steel end dome assemblies

The purpose of the cold mass assembly is to maintain the collared coil assembly at its nominal operating temperature of 1.9 K and to act as the transport mechanism for liquid helium between magnets when they are installed at CERN. Under normal operating conditions, the temperature of the vessel is 1.9 K with an internal pressure of 4.4 psig [1.3 bar].

The cold mass must satisfy all the requirements of the “Pressure Vessels” section (section 5031) of the Fermilab ES&H Manual. This section states that all applicable vessels shall adhere to the requirements of the ASME Boiler and Pressure Vessel Code Section, VIII.

This vessel will not be an ASME code stamped vessel. The intent of the design is to address and adhere to as many requirements of the ASME code as possible.

The assembly can be seen in Figure 2.0.1.

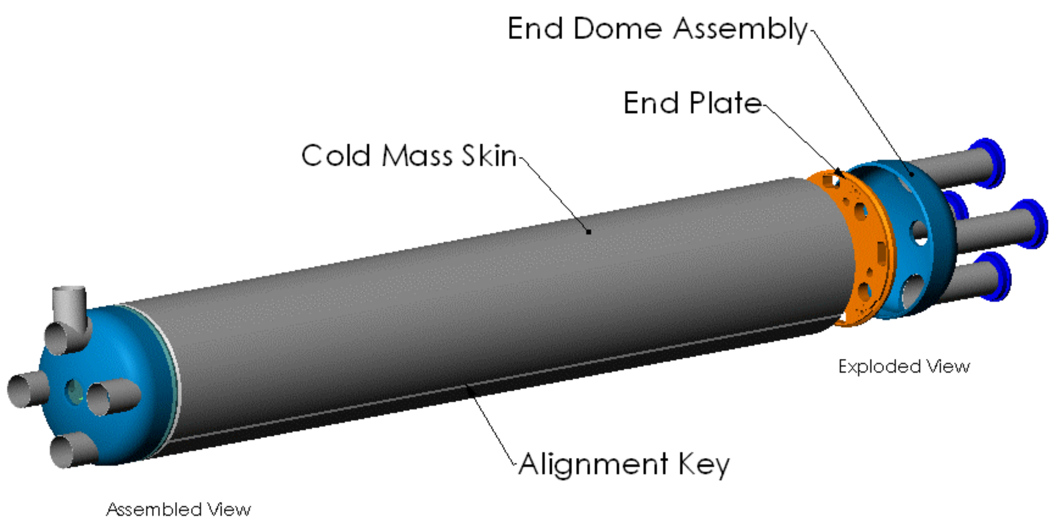

Figure 2.0.1 Cold Mass Assembly

The maximum stress that is allowed by Section II, Part D, Table 1A of the Code is as follows:

304 stainless steel: 20,000 psi

304L stainless steel: 16,700 psi.

Section 5031 of the Fermilab ES&H Manual requires de-rating of the allowable stress to 80% of the allowed value in cases where the vessel is either fabricated in-house or is not code-stamped. This reduces the allowed stress in pressure vessel calculations to the following:

304 stainless steel: 16,000 psi

304L stainless steel: 13,360 psi.

The design pressure for the LHC IR Quadrupoles for CERN is 290 psi. It was discovered after procurement of the dome that the dome is not properly sized for this service. The reinforcement requirement for the openings in the dome was not satisfied and this is shown in Section 2.3.1. Rather than add reinforcement to the dome, a decision was made to de-rate the MAWP of the prototype cold mass assembly, Q2P1. This was acceptable because the magnet will only be used for testing purposes at Fermilab. The MAWP (design pressure) was found to be 175 psi and the determination of this value is shown in Section 2.3.1. Should a quench occur on the test stand, there is no risk of over pressurizing the cold mass since the feedbox at MTF is rated for 100 psi and has a relief set at or below this value.

2.1 Cold mass skin and alignment key weld

The cold mass skin consists of two half shells welded longitudinally together with an alignment key. A cross section through the cold mass is shown in Figure 2.1.1.

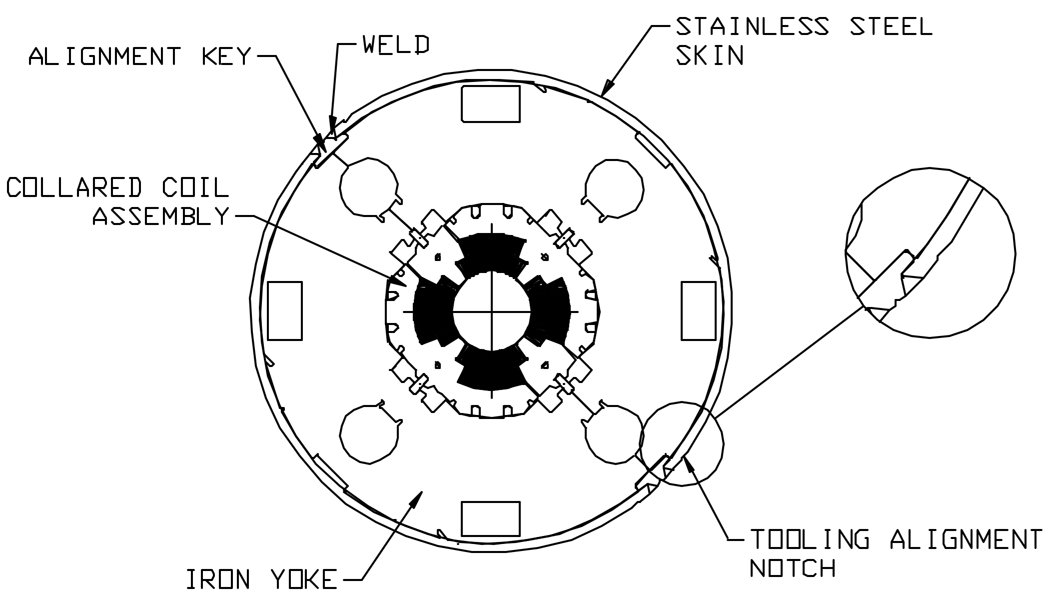

Figure 2.1.1

The alignment key weld is a Category A, Type 2 weld as described in UW-3 and UW-12 of the Code. It is a single welded butt joint with a backing strip and no radiographic examination and therefore the joint efficiency, E, is 0.65. The design pressure used in this calculation is 175 psi as discussed in Section 2.0.

The minimum thickness requirement is given by UG-27 and is the larger of:

$$t = \frac{PR}{SE - 0.6P}$$

or

$$t = \frac{PR}{2SE + 0.4P}$$

where: P = internal design pressure = 175 psi

R = inside radius of shell = 7.874 inches

S = allowable material stress = 13,360 psi

E = joint efficiency = 0.65

For this case, t = 0.161 inches is the larger value. The minimum skin thickness is 0.26 inches, which is located at the tooling alignment notch as shown in Figure 2.1.1, so this requirement is satisfied.

2.2 Cold mass skin to end plate weld

The cold mass skin to end plate weld conforms to UW-13.2 (d) of the Code and is shown in Figure 2.2.1.

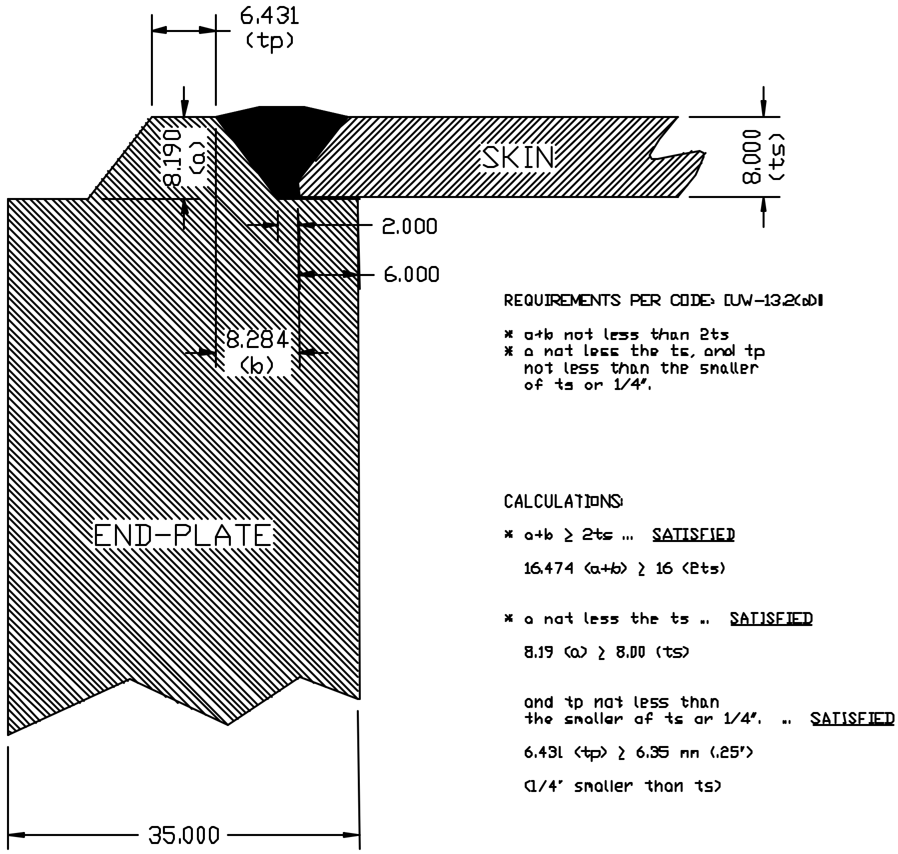

Figure 2.2.1 Detail of cold mass skin to end plate weld.

Using the notation from the figure:

$a = 8.190$ mm

$b = 8.284$ mm

$t_s = 8.00$ mm

$t_p = 6.43$ mm

UW-13.2 (d) requires that:

(1) $a+b \geq 2t_s$

(2) $a \geq t_s$

(3) $t_p \geq t_s$ or $t_s \geq$ ¼ in (6.35 mm)

All three requirements are satisfied. Technically, Figure UW13.2 (d) applies to a shell welded to a pressure head. In this design, the end plate is not a pressure head but the weld was designed as such.

2.3 End dome Assembly

The end dome assembly is attached to both ends of the cold mass to create the helium containment vessel. There are pipes attached to openings in the dome as shown in Figure 2.3.1. These pipes transport helium as well as provide a feedthrough for the wiring between the magnet and the feedbox when installed at MTF.

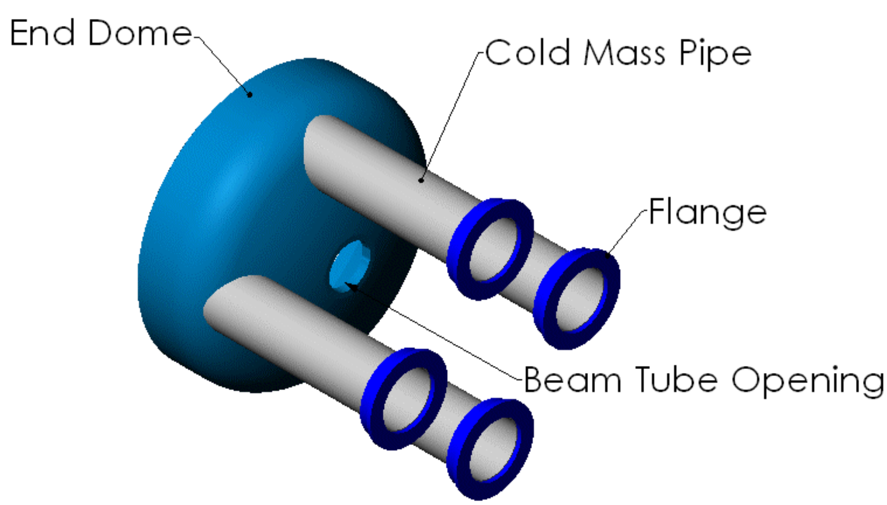

Figure 2.3.1 End Dome Assembly

2.3.1 End dome

The end dome is a formed ellipsoidal head. The minimum required thickness is given by UG-32 (d)

$$t = \frac{PD}{2SE - 0.2P}$$

where:

t = minimum required thickness of head after forming, inches

P = internal design pressure = 290 psi

D = inside length of the major axis (ID) = 15.628 inches

S = allowable material stress = 16,000 psi

E = joint efficiency = 0.60

Calculating:

t = [290(15.628)]/[(2*16000*.06)-0.2*290] = 0.237 inches

This requirement is satisfied.

This is the minimum required thickness for the end dome without any openings. Since there are openings in the dome, the requirement for reinforcement must be checked. There are five openings in the dome so the requirement for reinforcement is given by UG-42 of the Code, "Reinforcement of Multiple Openings". The center opening is for the beam tube. The other four openings are for the cold mass pipes. When considering the required reinforcement, Section UG-42 (a) (3) states "A series of openings all on the same center line shall be treated as successive pairs of openings." From this statement and the symmetry in the hole pattern, only two adjacent openings need to be addressed. These are the center opening and one of the four pipe openings. See Figure 2.3.1.1.

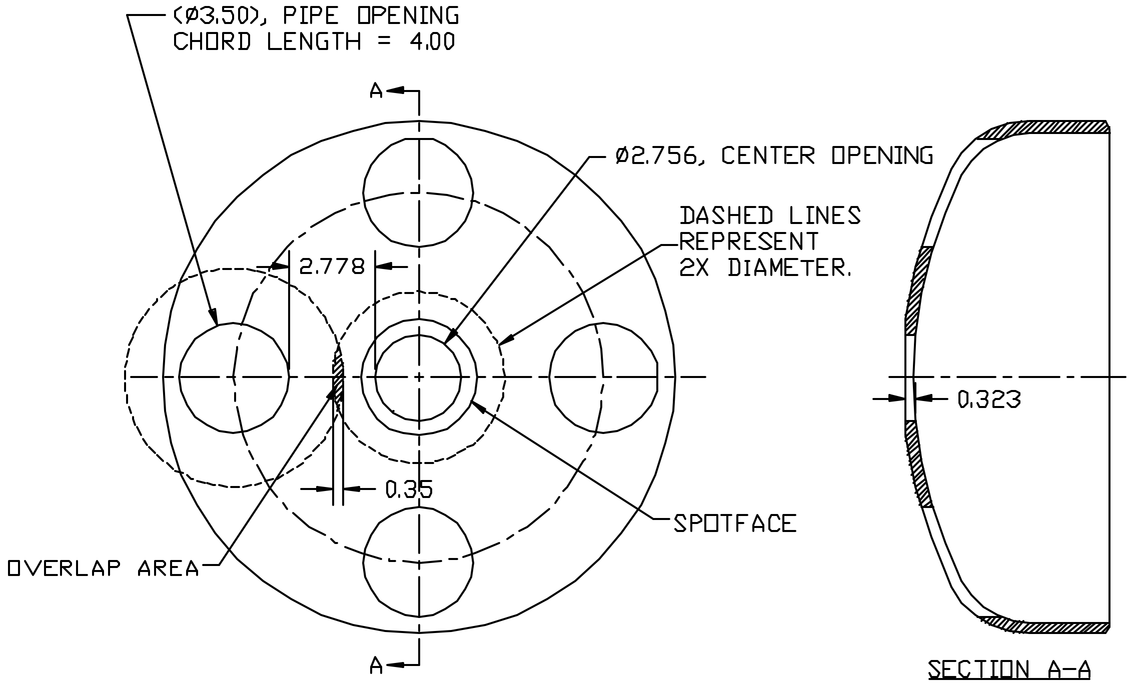

Figure 2.3.1.1 Detail of dome.

Section UG-37 of the Code requires that the minimum area of reinforcement for these openings is:

$$A_r = dt_rF + 2t_nt_rF(1 - f_{r1})$$

where: A_{r1} = are required for center hole

A_{r2} = are required for pipe hole

d_1 = inside diameter of center opening = 2.756 inches

d_2 = inside diameter (chord length across opening) of pipe opening = 4.00 inches

t_r = minimum required thickness of the shell = 0.237 inches

F = correction factor = 1

t_{n1} = nozzle wall thickness for center opening = 0.157 inches

t_{n2} = nozzle wall thickness for pipe opening = 0.065 inches

f_{r1} = strength reduction factor = 1

For this case, $A_{r1} = 0.653\ in^2$ and $A_{r2} = 0.947\ in^2$.

The area for reinforcement available in the dome is given by the larger of:

$$A_{ac} = d(E_1 t - Ft_r) - 2t_n(E_1 t - Ft_r)(1 - f_{r1})$$

or

$$A_{ac} = 2(t + t_n)(E_1 t - Ft_r) - 2t_n(E_1 t - Ft_r)(1 - f_{r1})$$

where: A_{ac1} = calculated required area for the center opening

A_{ac2} = calculated required area for the pipe opening

E_1 = weld efficiency = 1

t = dome thickness as-built = 0.375 inches

t_{co} = dome thickness at spotface, thickness used for center opening = 0.323 inches

There is a spotface on the end of the dome which reduces the thickness of the dome to 0.323 inches. This thickness is used in the calculation for the available area for the center opening. The larger values for each area are found to be: $A_{ac1} = 0.238\ in^2$ and $A_{ac2} = 0.553\ in^2$ from the two expressions above.

It can be seen that these openings are spaced at less than two times their average diameter. Section UG-42 (a) (1) of the Code requires that the available area between openings shall be proportioned between the two openings by the ratio of their diameters. The overlap area is given by:

$$A_{over} = (\text{ratio})\ L_{over}\ (t - t_r)$$

where: A_{over1} = overlap area of the center opening

A_{over2} = overlap area of the pipe opening

L_{over} = length of overlap = 0.35 inches

$ratio_1$ = ratio for center opening = $d_1/(d_1+d_2)$ = 0.41

$ratio_2$ = ratio for pipe opening = $d_2/(d_1+d_2)$ = 0.59

This gives A_{over1} = 0.019 in^2 and A_{over2} = 0.029 in^2. The overlap area from the center opening is subtracted from the available reinforcement area of the pipe opening and vice versa. This leads to the available area for each opening as follows:

$$A_{a1} = A_{ac1} - A_{over2} \qquad \text{and} \qquad A_{a2} = A_{ac2} - A_{over1}$$

This results in the true available reinforcement area for each opening: A_{a1} = 0.209 in^2 and A_{a2} = 0.534 in^2. This shows that the available area is less than the required area for reinforcement. This requirement is not satisfied.

Rather than add reinforcement to the dome, the MAWP of the dome, and therefore the cold mass, is de-rated from the original design pressure of 290 psi. The new value for the design pressure and the MAWP is 175 psi. Finding this value was an iterative process. Various values of pressure were input until the maximum value for the pressure was obtained. The final result is shown below.

The required thickness is given by UG-32 (d):

$$t_r = \frac{PD}{2SE - 0.2P}$$

where:

t_r = minimum required thickness of head after forming, inches

P = internal design pressure (de-rated value)= 175 psi

D = inside length of the major axis (ID) = 15.628 inches

S = allowable material stress = 16,000 psi

E = joint efficiency = 0.60

This gives the result, t_r = 0.143 inches.

Section UG-37 of the Code requires that the minimum area of reinforcement for these openings is:

$$A_r = dt_rF + 2t_nt_rF(1 - f_{r1})$$

where: A_{r1} = are required for center hole

A_{r2} = are required for pipe hole

d_1 = inside diameter of center opening = 2.756 inches

d_2 = inside diameter (chord length across opening) of pipe opening = 4.00 inches

t_r = minimum required thickness of the shell = 0.143 inches

F = correction factor = 1

t_{n1} = nozzle wall thickness for center opening = 0.157 inches

t_{n2} = nozzle wall thickness for pipe opening = 0.065 inches

f_{r1} = strength reduction factor = 1

For this case, $A_{r1} = 0.393$ in^2 and $A_{r2} = 0.571$ in^2.

The area for reinforcement available in the dome is given by the larger of:

$$A_{ac} = d(E_1t - Ft_r) - 2t_n(E_1t - Ft_r)(1 - f_{r1})$$

or

$$A_{ac} = 2(t + t_n)(E_1t - Ft_r)-2t_n(E_1t - Ft_r)(1 - f_{r1})$$

where: A_{ac1} = calculated required area for the center opening

A_{ac2} = calculated required area for the pipe opening

E_1 = weld efficiency = 1

t = dome thickness as-built = 0.375 inches

t_{co} = dome thickness at spotface, thickness used for center opening = 0.323 inches

There is a spotface on the end of the dome which reduces the thickness of the dome to 0.323 inches. This thickness is used in the calculation for the available area for the center opening. The larger values for each area are found to be: $A_{ac1} = 0.497$ in^2 and $A_{ac2} = 0.929$ in^2 from the two expressions above.

It can be seen that these openings are spaced at less than two times their average diameter. Section UG-42 (a) (1) of the Code requires that the available area between openings shall be proportioned between the two openings by the ratio of their diameters. The overlap area is given by:

$$A_{over} = (\text{ratio})\ L_{over}\ (t - t_r)$$

where: A_{over1} = overlap area of the center opening

A_{over2} = overlap area of the pipe opening

L_{over} = length of overlap = 0.35 inches

ratio_1 = ratio for center opening = $d_1/(d_1+d_2) = 0.41$

ratio_2 = ratio for pipe opening = $d_2/(d_1+d_2) = 0.59$

This gives $A_{over1} = 0.033$ in^2 and $A_{over2} = 0.049$ in^2. The overlap area from the center opening is subtracted from the available reinforcement area of the pipe opening and vice versa. This leads to the available area for each opening as follows:

$$A_{a1} = A_{ac1} - A_{over2} \qquad \text{and} \qquad A_{a2} = A_{ac2} - A_{over1}$$

This results in the true available reinforcement area for each opening: A_{a1} = 0.448 in^2 and A_{a2} = 0.897 in^2. The available areas are greater than the required areas, so this requirement is met.

Section UG-42 (2) requires that at least 50% of the required area of reinforcement must be between the two openings. The required area between the openings is given by:

$$A_{50\%R} = (A_{r1} + A_{r2})/2 = 0.482 \text{ in}^2.$$

The actual area available between openings is given by:

$$A_{between} = L_{between}\,(t - tr)$$

where: $L_{between}$ = distance between openings = 2.778 inches

This gives $A_{beteween}$ = .645 in^2. This requirement is satisfied.

All the requirements of the Code have been satisfied for an internal pressure of 175 psi. This value is the MAWP for the Q2P1 cold mass.

2.3.2 Cylindrical section

The dome consists of the elliptical portion as well as a small straight cylindrical section. This can be seen in Figure 2.3.1.1. This section is treated as cylindrical shell and the required thickness is given by UG-27 of the Code. The minimum thickness is given by the larger of:

$$t = \frac{PR}{SE - 0.6P}$$

or

$$t = \frac{PR}{2SE + 0.4P}$$

where: P = internal design pressure = 175 psi

R = inside radius of shell = 7.874 inches

S = allowable material stress = 16,000 psi

E = joint efficiency = 0.60

For this case, t = 0.145 inches is the larger value. The shell thickness is 0.375 inches so this requirement is satisfied.

2.3.3 Cold mass pipes

The welds between the cold mass pipes and the end dome are all single sided welds made on the outside of the dome. The maximum load on the weld is a combined load due to the internal pressure and the attached bellows. The force due to internal pressure is equal to the cross sectional area in the pipe multiplied by the design pressure. Using the design pressure of 175 psi, this force is equal to 1,600 lbs. The bellows has an axial spring constant of 68 lb/in and a maximum travel of 1.67 inches. This results in a bellows force of 114 lbs. This force is combined with the force due to internal pressure for a combined force of 1,714 lbs.. As shown in Fermilab drawings 5520-MD-390053 and 5520-MD-390054, these welds are specified to be a 2 mm (0.08 inch) fillet weld. The stress on the weld is given by

$$t_w = \frac{f_a}{(l)(t_w)}$$

where: τ_w = shear stress in the weld

f_a = axial force = 1,714 lb

l = linear length of weld = 11.0 inches

t_w = weld equivalent thickness = 2 mm/√2 = 1.414 mm = 0.056 inch

For this case, the weld stress, τ_w, is 2,782 psi which is well below that allowed by UW-15 of the Code given by:

$$(20{,}000 \text{ psi})(0.8)(0.49) = 7{,}840 \text{ psi}$$

The welds between the cold mass pipes and the end flanges are category C lap welds as described in UW-3 (a) (2) and UW-9 (e) of the Code. UW-9(e) requires that the overlap be not less than four times the thickness of the inner plate. In the case of the cold mass pipe, the tube thickness is 0.065 inch. The overlap at the end flanges is 0.69 inch so the requirement is met. The only load acting on this flange is an axial load from the maximum design pressure of 175 psi. The total axial force acting on the flange is 1,600 lb. At the end flange, this force is resisted by the weld between the cold mass pipe and the end flange. As shown on Fermilab drawing 5520-MD-390053, this weld is specified to be a 2 mm (0.08 inch) fillet. The stress on the weld is given by

$$s_w = \frac{f_a}{(l)(t_w)}$$

where: σ_w = stress in the weld

f_a = axial force = 1,600 lb

l = linear length of weld = 11.0 inches

t_w = weld equivalent thickness = 2 mm/√2 = 1.414 mm = 0.056 inch

For this case, the weld stress, σ_w, is 2,600 psi which is well below that allowed by UW-18 of the Code given by:

$$(20{,}000 \text{ psi})(0.8)(0.55) = 8{,}800 \text{ psi}$$

2.3.4 Beam tube

The beam tube is inserted through the center of the cold mass. The internal pressure of the cold mass acts as an external pressure on the beam tube. The thickness of a shell or tube under external pressure is given by section UG-28 of the Code. Following the Code steps:

(c) *Cylindrical Shells and Tubes*. The required minimum thickness of a cylindrical shell or tube under external pressure, either seamless or with longitudinal butt joints, shall be determined by the following procedure.

(1) *Cylinders having* D_0/t *values* ³ *10*:

Step 1. Assume a value for t and determine the ratio L/D_0 and D_0/t.

For this case t=1.75 mm, D_0=66.7 mm and L=11 m.
Then, L/D_0>50 and D_0/t=38.1.

Step 2. Enter Fig. G in Subpart 3 of Section II, Part D of the Code at the value of L/D_0 determined in Step 1. For values of L/D_0 greater than 50, enter the chart at a value of L/D_0=50.

Step 3. Move horizontally to the line for the value of D_0/t determined by Step 1. Interpolation may be made for intermediate values of D_0/t. From this point of intersection move vertically downward to determine the value of factor A. From the chart A=0.000788

Step 4. Using the value A calculated in Step 3, enter the applicable material chart in Subpart 3 of Section II, Part D of the Code for the material under consideration. Move vertically to an intersection with the material/temperature line for the design temperature.

Step 5. From the intersection obtained in Step 4, move horizontally to the right and read the value of factor *B*.

From Fig. HA-4 for 316 L stainless steel, for A=0.000788 and for operation up to 100 F, B=8610.

Step 6. Using the value of *B*, calculate the value of the maximum allowable external working pressure P_a using the following formula:

$$P_a = \frac{4B}{3(D_o/t)}$$

This gives P_a=301.3 psi. This requirement is satisfied.

The weld between the beam tube and the flange is a category C lap weld as described in UW-3 (a) (2) and UW-9 (e) of the Code. UW-9(e) requires that the overlap be not less than four times the thickness of the inner plate. In the case of the beam tube, the tube thickness is 0.073 inch. The overlap at the end flanges is 0.73 inch so the requirement is met. The only load acting on this flange is an axial load from the maximum design pressure of 175 psi. The total axial force acting on the flange is 116 lb. At the flange, this force is restricted by the weld between the beam tube and the flange and also by the weld between the flange and the end dome. It will be assumed that only one weld is resisting the load and the smaller weld will be chosen. This is the weld between the beam tube and the flange. The stress on the weld is given by

$$s_w = \frac{f_a}{(l)(t_w)}$$

where: σ_w = stress in the weld

f_a = axial force = 116 lb

l = linear length of weld = 4.16 inches

t_w = weld equivalent thickness = 1.8 mm/$\sqrt{2}$ = 1.27 mm = 0.050 inch

For this case, the weld stress, σ_w, is 558 psi which is well below that allowed by UW-18 of the Code given by:

$$(20{,}000 \text{ psi})(0.8)(0.55) = 8{,}800 \text{ psi}$$

2.4 End dome to end plate weld

The end dome to end plate weld conforms to ASME Code, UW-13.2 (d) and is shown in Figure 2.4.1.

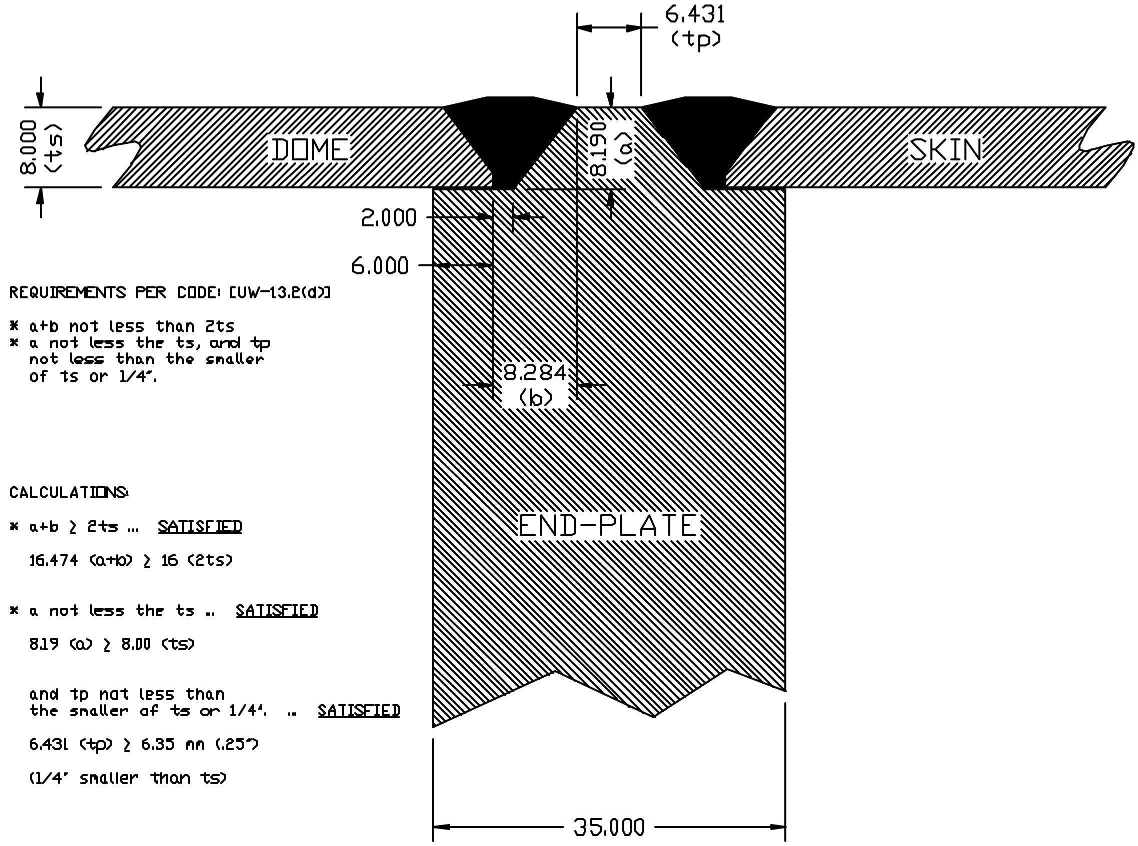

Figure 2.4.1 Detail of cold mass skin to end plate weld.

Using the notation from the figure:

a = 8.190 mm

b = 8.284 mm

t_s = 8.00 mm

t_p = 6.43 mm

UW-13.2 (d) requires that:

(1) $a+b \geq 2t_s$

(2) $a \geq t_s$

(4) $t_p \geq t_s$ or $t_s \geq$ ¼ in (6.35 mm)

All three requirements are satisfied.

2.5 Non-pressure loads

The primary stress in the cold mass assembly is due to internal pressure. There are, however, other stress inducing loads that must be addressed. These are discussed in sections 2.5.1 and 2.5.2.

2.5.1 Welding and cooldown

There is stress in the cold mass skin due to shrinkage that occurs during welding and cooldown and due to mechanical support of the magnet in the vacuum vessel. This can be broken down as follows: 14,550 psi due to initial welding, 29,020 psi due to cooldown, 3,600 psi due to welding the attachment lugs, and 900 psi acting at the supports (see 2.5.2) giving a total combined stress at operating temperature of 48,070 psi (331 MPa) The initial weld-induced stress was measured during construction using strain gages mounted directly to the skin. The cooldown stress is estimated by calculation. This stress is constant at the operating temperature regardless of internal pressure and is not considered by Division 1 of the Code. Division 2 of the Code considers this a secondary stress(1). The allowable stress is given by Paragraph 4-134 of Division 2:

$$\sigma_a = 3\sigma_m f$$

where: σ_a = allowable stress

σ_m = stress from Sec. 2, Part D, Table 1A for 304L S.S. = 16,700 psi

f = Fermilab de-rating factor = 0.8

[1] "Classification of Stresses For The Skin Of The Cold Mass", Robert L. Cloud & Associates, Inc. prepared for the SSC.

For this case, σ_a = 40,080 psi. The stress in the skin of 48,070 is higher than that allowed by Division 2 of the Code.

The cold mass attachment lugs are welded to the cold mass skin at four places using a two-pass weld. It is difficult to assess the stress resulting from this weld, however, the strain gauge data from the skin-to-alignment key weld showed a maximum stress of 25 MPa (3,600 psi) after two passes. Using this, it should be safe to assume a maximum additional skin stress of 3,600 psi immediately adjacent to the lugs. This stress is included in the total combined stress referenced above.

The internal pressure that would correspond to the average shell pre-stress developed during welding and cooldown is given by Division 1 , UG-27 (1):

$$P = \frac{s_w E t}{R + 0.6t}$$

where: P = internal pressure

σ_w = 48,070 psi, the average stress in skin during welding & cooldown,

E = joint efficiency = 0.65

t = skin thickness = 0.26 inches

R = inside radius of shell = 7.874 inches

For this case, the internal pressure is 1,010 psi, i.e. an internal pressure of 1,010 psi would be required to increase the shell stress above the assembly and cooldown stress of 48,070 psi. The design pressure of 175 psi is well below this value.

2.5.2 Gravity Load

There are two sources of gravity loads on the cold mass skin. The dead weight load acting on the cold mass at the support points produces bending stresses in the cold mass skin. These stresses are a maximum at the supports. The maximum bending stress is 1220 psi assuming a simply supported structure. These stresses are longitudinal and are therefore not additive to the circumferential stress that limits the internal pressure. There is also a bending moment at each of the support lugs. The prototype cold mass weighs 13,500 lb. There are four lugs, each of which is 12 inches long. The resulting force on the welds is 280 lb/inch of weld. This force produces a skin stress of 900 psi directly under the weld. There are no welds through the thickness of the material at any of these locations. This stress is included in the total combined stress referenced in 2.5.1.

2.6 Pressure Testing

The cold mass assembly will be pressure tested in accordance with Section 5034 of the Fermilab ES&H Manual and UG-100 of the Code. The test pressure is 220 psi, which is 1.25 times the design pressure. The test will be performed after normal working hours and only personnel directly involved with the test will be present. The test medium will be gaseous nitrogen.

2.7 Summary

The LHC cold mass satisfies all the requirements of the ASME Code, however, it falls short of satisfying the requirements of the Fermilab ES&H Manual due to the 20% allowed stress derating. It was shown that the MAWP for the prototype cold mass, as defined by the Code is 175 psi.

Chapter 3

LHC Interaction Region Quadrupole

Cryogenic piping

3.0 Introduction

The cryogenic pipes perform a variety of functions. They transport cryogens down the length of the cryostat during cooldown, warm-up, and in operation. There are nine distinct tube that comprise the cryogenic piping. Their descriptions and a summary of their operating parameters are shown in table 3.0.1.

Table 3.0.1. Cryogenic piping parameters

Description	Fluid	OD (mm)	ID (mm)	P oper (bar)	P max (bar)	T (approx)	Flow (g/s)
Pumping line	Ghe	88.90	85.60	0.016	4.0	1.8 K	8.6
Heat exchanger outer shell	Lhe	168.28	162.74	1.3	20.0	1.9 K	0.0
Heat exchanger inner tube	Lhe	97.54	96.01	0.016	4.0	1.8 K	8.6
Cooldown line	Lhe	44.45	41.96	1.3	20.0	1.9 K	30.0
LHe supply *	Lhe	15.88	13.39	0.016	4.0	1.8 K	8.6
4.5K supply *	Lhe	19.05	15.75	1.3	20.0	4.5 K	1.1
4.5K return *	Lhe	19.05	15.75	1.3	20.0	4.5 K	1.1
50-70K shield supply	Ghe	38.10	31.75	19.5	22.0	60 K	5.0
50-70K shield return	GHe	38.10	31.75	19.0	22.0	65 K	5.0

*: Not used on prototype magnets, Q2P1 or Q2P2.

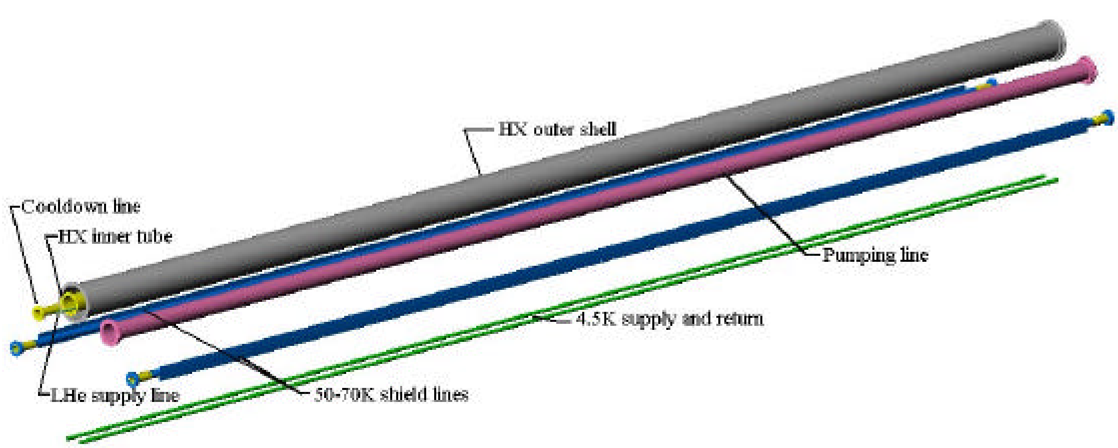

Figure 3.0.1 LHC cryostat cryogenic piping

Figure 3.0.1 illustrates all the cryogenic lines in an LHC cryostat. Note that as noted in table 3.0.1, the prototypes do not contain all of these lines.

3.1 Design codes and evaluation criteria

The LHC quadrupole cryostat piping must satisfy all the requirements of the "Pressure Piping Systems" section (section 5031.1) of the Fermilab ES&H Manual. This section states that all applicable pressure piping systems shall adhere to the requirements of the ASME B31 code series. ASME B31.3, "Chemical Plant and Petroleum Refinery Piping", is used for the analyses presented here. There is no part in the B31 series directly applicable to cryogenic piping.

3.2 Materials

The pumping line, heat exchanger outer shell, cooldown line, Lhe supply, and 4.5K supply and returns are fabricated from 304 stainless steel. The heat exchanger inner tube is an OFHC copper corrugation. The 50-70K shield supply and return are 6063-T5 aluminum extrusions.

3.3 Pressure loading and analysis

With the exception of the heat exchanger outer shell and inner tube, the minimum thickness is evaluated using the procedures in 304.1.2(a) of ASME B31.3. The minimum tube thickness for seamless or longitudinally welded piping for $t<D/6$ is given by:

$$t = \frac{PD}{2SE}$$

where: t = wall thickness

P = internal design pressure

D = outside diameter

S = allowable stress from table A-1

E = quality factor from table A-1A or A-1B

Table 3.3.1 summarizes the results of the wall thickness calculation for each of the applicable lines.

Table 3.3.1. Cryogenic piping parameters

Description	P (psi)	D (in)	S (psi)	E	t req'd (in)	P at MTF (psi) **	t actual (in)
Pumping line	59	3.50	20,000	1.0	0.005	1	0.065
Heat exchanger outer shell	na (see below)					100	0.109
Heat exchanger inner tube	na (see below)					1	0.310
Cooldown line	294	1.75	20,000	1.0	0.013	100	0.065
LHe supply *	59	0.63	20,000	1.0	0.001	1	0.065
4.5K supply *	294	0.75	20,000	1.0	0.006	100	0.065
4.5K return *	294	0.75	20,000	1.0	0.006	100	0.065
50-70K shield supply	323	1.50	7,300	1.0	0.033	100	0.125
50-70K shield return	323	1.50	7,300	1.0	0.033	100	0.125

*: Not used on prototype magnets, Q2P1 or Q2P2.

**: Relief valve setting at MTF.

In all cases the actual wall thickness of the piping is greater than the minimum required by ASME B31.3. Also in all cases, the maximum pressures at MTF as established by the relief valve settings are less than the design pressures.

3.3.1 Heat exchanger outer shell

The outer shell of the external heat exchanger is a special case when considering the cryogenic piping because it is over 6 inches in diameter, i.e. the diameter above which the boiler and pressure vessel code applies, not the piping code. Application of the Code to determine the minimum required thickness for the outer shell yields the results shown in table 3.3.1.1.

Table 3.3.1.1 Outer shell as a pressure vessel (governing equations (UG-27(c))

$$t = \frac{PR}{SE - 0.6P}(circumferential stress) \text{ or } t = \frac{PR}{2SE + 0.4P}(longitudinal stress)$$

Variable	Value	Units	Descriptions and References
P	300	psi	Internal design pressure
R	3.125	in	Shell inside radius
S	16000	psi	Subpart 1, Section II, Part D, Table 1A, derated to 80% of allowed
E	0.70		Weld joint efficiency (Table UW-12)
t(c)	0.085	in	Minimum shell thickness when sized for circumferential stress
t(l)	0.042	in	Minimum shell thickness when sized for longitudinal stress
t	0.085	in	Minimum shell thickness

For this case the minimum wall thickness required is 0.085 inch. The outer shell of the heat exchanger is 6 inch IPS, schedule 5 with an outside diameter of 6.625 inches and a wall thickness of 0.109 inch so the requirement is satisfied.

3.3.1.1 End flanges

The welds between the tube ends and the end flanges are category C lap welds as described in UW-3(a)(2) and UW-9(e) of the Code. UW-9(e) requires that the overlap be not less than four times the thickness of the inner plate. In the case of the heat exchanger outer shell, the tube thickness is 0.109 inch. The overlap at the end flanges is 0.6 inch so the requirement is met. The only load acting on this flange is an axial load from the maximum design pressure of 300 psi. The total axial force acting on the flange is 9,700 lb. At the end flange, this force is resisted by the weld between the outer shell and the end flange. As shown on Fermilab drawing 5520-ME-390002, this weld is specified to be a 3 mm (0.12 inch) fillet. The stress on the weld is given by

$$s_w = \frac{f_a}{(l)(t_w)}$$

where: σ_w = stress in the weld

f_a = axial force = 9,700 lb

l = linear length of weld = 20.8 inches

t_w = weld equivalent thickness = 3 mm/$\sqrt{2}$ = 2.12 mm = 0.084 inch

For this case, the weld stress, σ_w, is 5,550 psi which is well below that allowed by UW-18 of the Code given by:

$$(20{,}000 \text{ psi})(0.8)(0.55) = 8{,}800 \text{ psi}$$

3.3.1.2 Cold mass connection

The heat exhanger outer shell connects to the cold mass through a short vertical tube. The connection of this tube to the outer shell constitutes an opening in the vessel that potentially needs reinforcement. Section UG-37 of the Code requires that the minimum area of reinforcement for these openings is:

$$A_r = dt_rF + 2t_nt_rF(1 - f_{r1})$$

where: A_r = area required

d = inside diameter of opening = 3.17 inches

t_r = minimum required thickness of the shell at the design pressure computed using UG-27(c)(1) = 0.048 (see table 3.3.1)

F = correction factor = 1

t_n = nozzle wall thickness = 3 mm = 0.12 inches

f_{r1} = strength reduction factor = 1

For this case, A_r = 0.15 in^2. The area for reinforcement available in the shell is given by the larger of:

$$A_1 = d(E_1t - Ft_r) - 2t_n(E_1t - Ft_r)(1 - f_{r1})$$

or

$$A_1 = 2(t + t_n)(E_1t - Ft_r)-2t_n(E_1t - Ft_r)(1 - f_{r1})$$

where: E_1 = weld efficiency = 1

t = vessel wall thickness = 0.109 inches

For this case, $A_1 = 0.19$ in^2 from the two expressions above. Since the available area, A_1 is greater than the required are A_r, no additional reinforcement is necessary.

The cold mass connection tube is attached to the outer shell using a fillet weld. The only load acting on this joint is an axial load from the maximum design pressure of 300 psi. The total axial force acting on the flange is 2,500 lb. The maximum force which can be supported by these welds is given by:

$$F = s_w pdt_w E$$

where: F = maximum allowed force in the weld

σ_w = stress in the weld = (20,000 psi)(0.8) = 16,000 psi

d = effective diameter of the weld = 3.5 inches

t_w = weld equivalent thickness = 3 mm/$\sqrt{2}$ = 2.12 mm = 0.084 inch

E = weld efficiency = 49% (per UW-15)

For this case, F = 7,240 lb so the weld is sufficient.

3.3.2 Heat exchanger inner tube

The inner tube of the external heat exchager is an OFHC copper corrugated tube. The dimensions are shown on Fermilab drawing 5520-MC-390011. Copper is required for thermal conductivity. The corrugations provide some reservoir for liquid and add to the structural strength of the tube. The pressure loading is shown in table 3.0.1. The most significant load is an external pressure at 20 bar or 300 psi.

The piping code doesn't explicitly address corrugated tubes. It addresses metal expansion joints, but this tube doesn't fall into that category. As a result, a finite element model of the tube was created and subjected to 60 psi internal and 300 psi external pressue loads. The results from these two analyses are shown in figures 3.3.2.1 and 3.3.2.2.

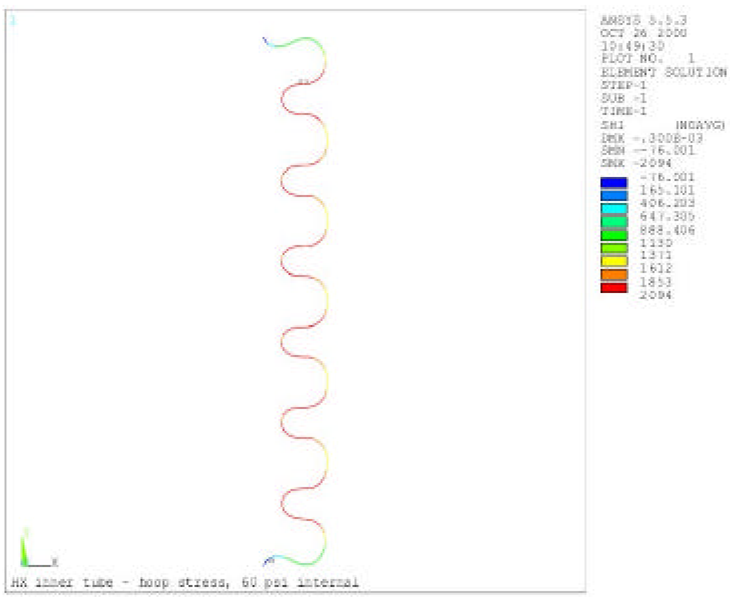

Figure 3.3.2.1   Heat exchanger inner tube hoop stress at 60 psi internal pressure

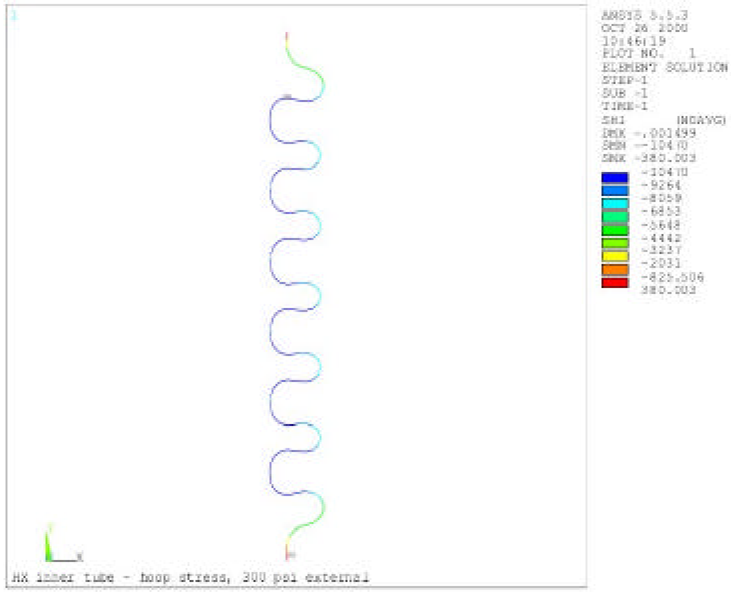

Figure 3.3.2.2 Heat exchanger inner tube hoop stress at 300 psi external pressure

In addition a sample of the inner tube was subjected to a hydrostatic test in a fixture made specifically for that purpose. The sample was tested to 500 psi external pressure with no visible distortion of the convolutions.

3.4 Summary

The LHC prototype cryostat cryogenic piping satisfies all requirements of the ASME Boiler and Pressure Vessel Code, ASME B31.3, and the Fermilab ES&H manual with the exception of the heat exchanger inner tube which is not explicitly addressed by the codes. However, in the case of this tube, analysis and test results indicate that the integrity of this tube is not compromised by any operating mode at MTF.

Chapter 4

LHC Interaction Region Quadrupole

Vacuum vessel

4.0 Introduction

The functions of the vacuum vessel are to contain the magnet's insulating vacuum and to provide the structural support of the magnet, shield, and internal piping to the accelerator tunnel floor. In operation, the vacuum vessel is pressurized externally with a differential pressure of one atmosphere. In the event of an internal piping failure the vessel may become pressurized internally. The maximum allowable working pressure is determined in this chapter for both internal and external pressure loading.

4.1 Design codes and evaluation criteria

The LHC quadrupole cryostat vacuum vessel must satisfy all the requirements of the "Vacuum Vessel Safety" section (section 5033) of the Fermilab ES&H Manual. This section states that all vacuum vessels shall adhere to the requirements of the Code where practical in a laboratory environment. Because the vacuum vessel contains cryogen lines, the potential for pressurization does exist. If one of these lines were to fail, cryogens could expand to pressurize the vessel to the vacuum system relief valve pressure of 1 psi. Both the Code and the ES&H Manual say that a vessel with an internal pressure of 15 psi or less is not considered a pressure vessel. Therefore, for the purposes of testing at MTF, the vessel functions strictly as a vacuum vessel.

4.2 Materials

The prototype vacuum vessel shell is fabricated from SA-516, grade 70 carbon steel. Material certifications are included in Appendix C. Flanges and access ports are fabricated from 304 stainless steel. The maximum stress allowed by the Code for SA-516, grade 70 is 20,000 psi and the allowable temperature range at this allowed stress is -20 to +500 °F (Section II, Part D, Subpart 1, Table 1A). Section 5031 of the Fermilab ES&H Manual requires derating of the allowable stress to 80% of the allowed value in cases where the vessel is either fabricated in-house or is not code-stamped. This reduces the allowed stress in pressure vessel calculations to 16,000 psi and corresponds to a safety factor of 5.

4.3 Structural loading and analysis

The mechanical load on the vacuum vessel consists of the gravity load of the internal components and the vessel itself, the internal radial vacuum load, and the axial vacuum load. The weight of each LHC cold mass is different so for the structural load due to gravity we will consider the weight of the heaviest assembly per unit length, Q3. The Q3 cold mass and internal components weigh 23,500 lb (10,680 kg) and are supported at two places along the length of the vacuum vessel. The radial vacuum load is equivalent to one atmosphere external pressure. The axial load is equal to the cross sectional area of the vessel times one atmosphere pressure or 15,000 lb (6,820 kg). Figure 4.3.1 illustrates a typical LHC IRQ quadrupole cryostat vacuum vessel. Attachments to the accelerator tunnel floor and the internal cold assembly are coincident and occur at the two reinforced sections. The end rings on either end of the vessel provide attachment points for vacuum bellows at one end and the turnaround can at MTF at the other. The four lugs shown at each end ring provide means for securing the vessel to the feedbox and turnaround can.

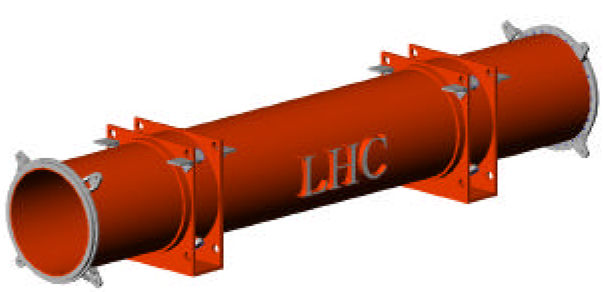

Figure 4.3.1 Typical LHC IRQ cryostat vacuum vessel

The stresses due to the gravity and vacuum loads were determined using a finite element model of the entire assembly. Figure 4.3.2 illustrates the finite element mesh. Gravity acts on the entire assembly. The vacuum loads are applied as a pressure of 15 psi acting inward on the outer vessel wall and as discreet forces acting along the length of the vessel and applied at the end ring.

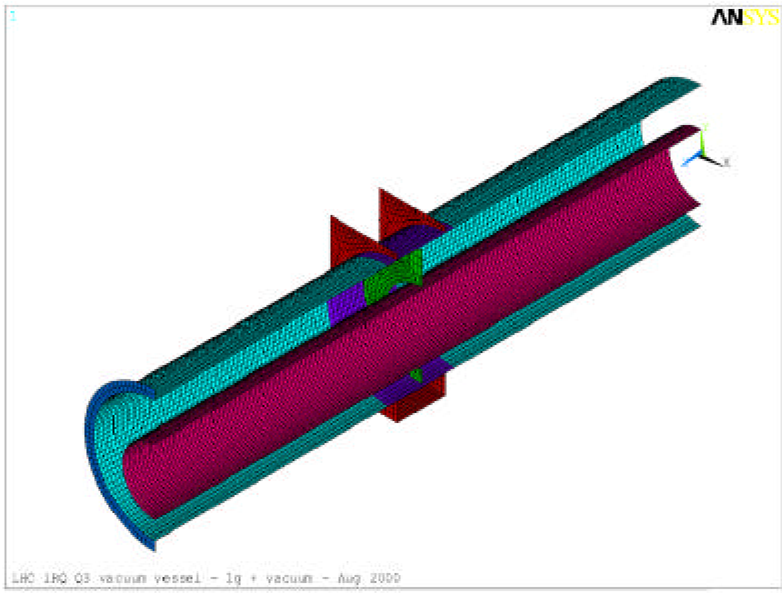

Figure 4.3.2 Finite element mesh from structural and vacuum load analysis

The stresses in the vacuum vessel wall resulting from these combined loads are shown in figure 4.3.3. The stress component displayed is the von Mises equivalent stress which is a combination of principal and shear stress components. It is commonly used to indicate the state of stress in structures that might be indicative of material yielding or failure. The maximum stress in the vessel shell from all the combined loads is 2335 psi and is a bending stress that occurs at the end of the vessel where the end ring attaches. This stress is below the allowed stress in the vacuum vessel material of 16,000 psi.

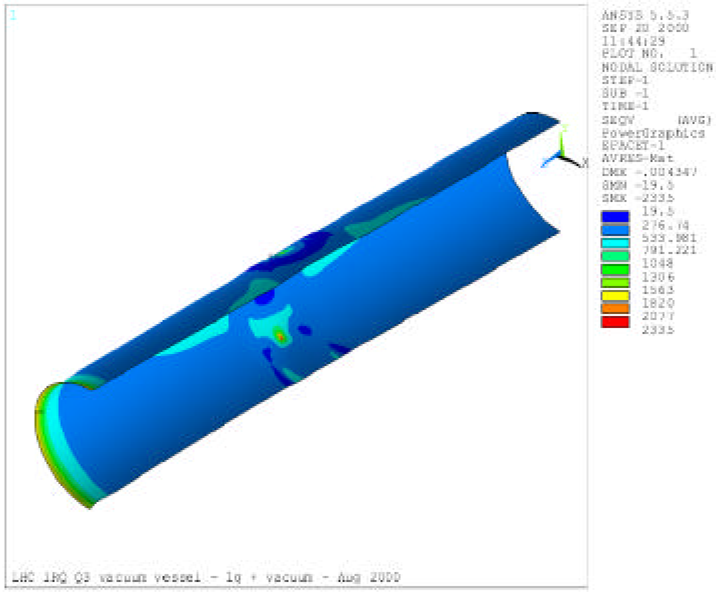

Figure 4.3.3 von Mises stress plot of the vacuum vessel shell only

4.4 Pressure loading and analysis

The vacuum vessel is fabricated in sections to allow adjustment of the individual pieces in an attempt to make as straight a vessel as possible. The individual tube sections are rolled and welded using full-penetration as shown in Fermilab drawing 5520-MC-390124. This drawing is typical of all vacuum vessel tube sections. The longitudinal seam weld is a double butt joint as described in UW-3(a)(1) and shown in table UW-12(1) of the Code. The weld joint efficiency, E, is 0.7 for the case where no radiographic examination is made. Tables 4.4.1 through 4.4.3 below summarize the Code calculations for the vacuum vessel as an externally pressurized vessel with 1 atmosphere external pressure and as a pressure vessel with 2 atmospheres internal pressure. The interconnecting sleeves and mounting frames are treated as stiffeners. The vacuum vessel has a pressure relief located on the MTF feedbox which opens just above atmospheric pressure. For the sake of the vacuum vessel acting as a pressure vessel however (table 4.4.3) the design pressure is defined to be 2 atmospheres.

Table 4.4.1 Shell as a vacuum vessel (governing equations (UG-28(c))

$$P_a = \frac{4B}{3(D_o/t)}(method1)\ or\ P_a = \frac{2AE}{3(D_o/t)}(method2)$$

Variable	Value	Units	Descriptions and References
Do	36.00	in	Vacuum vessel OD
L(total)	330.00	in	Total shell length
n	2		Number of stiffening rings
L	165.00		Distance between stiffeners
t	0.500	in	Vacuum vessel thickness
E	3.00E+07	psi	Young's modulus
L/Do	4.58		
Do/t	72		
A	0.00045		Subpart 3, Section II, Part D, Figure G.
B	6500		Subpart 3, Section II, Part D, Figure CS-1.
Pa (method 1)	120.37	psi	Calculated maximum allowable external working pressure
Pa (method 2)	125.00	psi	Calculated maximum allowable external working pressure

Table 4.4.2 Stiffening rings (governing equations (UG-29(a))

$$I_s = \frac{D_o^2 L_s (t + A_s/L_s) A}{14}\ and\ B = 3/4 \left(\frac{PD_o}{t + A_s/L_s}\right)$$

Variable	Value	Units	Descriptions and References
P	30	psi	External design pressure. Two atmospheres per FESHM 5033
Do	36.00	in	Vacuum vessel OD
Ls	165.00	in	Distance between stiffeners
t	0.500	in	Vacuum vessel thickness
As	29.528	in2	Assumed cross sectional area (38" OD, 1-1/2" wall, 19.685" long)
B	1193		UG-29, Step 1
A	7.95E-05		2 * B / E per UG-29 Step 5
Is	0.825	in4	Required stiffener I

Table 4.4.3 Shell as a pressure vessel (governing equations (UG-27(c))

$$t = \frac{PR}{SE - 0.6P}(circumferential stress)\ or\ t = \frac{PR}{2SE + 0.4P}(longitudinal stress)$$

Variable	Value	Units	Descriptions and References
P	30	psi	Internal design pressure
R	18.000	in	Shell inside radius
S	16,000	psi	Subpart 1, Section II, Part D, Table 1A, derated to 80% of allowed
E	0.70		Weld joint efficiency (Table UW-12)
t(c)	0.048	in	Minimum shell thickness when sized for circumferential stress
t(l)	0.024	in	Minimum shell thickness when sized for longitudinal stress
t	0.048	in	Minimum shell thickness

From table 4.4.1, the maximum allowable external working pressure of the vacuum vessel, Pa, is 120 psi. The minimum pressure required by the Fermilab ES&H manual, chapter 5033 is 2 atmospheres or 30 psi so the requirement is met. From table 4.4.2, the required section modulus of stiffeners is 0.825 in^4. The section modulus of the connecting rings is 5.5 in^4 so the requirement is met. Finally, from table 4.4.3, the minimum shell thickness for the vacuum vessel is 0.048 inch. The vessel wall is actually 0.5 inch so the requirement is met.

4.4.1 Connecting rings

The welds between the individual tube sections and the interconnecting sleeves are category C lap welds as described in UW-3(a)(2) and UW-9(e) of the Code. UW-9(e) requires that the overlap be not less than four times the thickness of the inner plate. In the case of the LHC vacuum vessels, the inner tube thickness is 1/2 inch. The overlap is 2 inches so the requirement is met. From the finite element analysis, the maximum stress in the tube section at the interconnecting sleeve is approximately 775 psi. Since the weld is not explicitly included in the finite element model it is necessary to scale the stress at the weld area by the ratio of the minimum thickness of the weld and the tube thickess. This gives:

$$s_w = s_t \frac{t_t}{t_w}$$

where: σ_w = stress in the weld

σ_t = stress in the tube = 775 psi (from finite element analysis)

t_t = tube thickness = 0.5 inch

t_w = weld equivalent thickness = 6 mm/$\sqrt{2}$ = 4.24 mm = 0.17 inch

For this case, the weld stress, σ_w, is 2,280 psi which is well below that allowed by UW-18 of the Code given by:

$$(20{,}000 \text{ psi})(0.8)(0.55) = 8{,}800\text{psi}$$

4.4.2 End flanges

The welds between the tube ends and the end flanges are category C laps weld as described in UW-3(a)(2) and UW-9(e) of the Code. UW-9(e) requires that the overlap be not less than four times the thickness of the inner plate. In the case of the LHC vacuum vessels, the inner tube thickness is 1/2 inch. The overlap at the end flanges is only 1 inch so the requirement is not met. The only load acting on this flange is an axial load from the internal vacuum. The total axial force acting on the flange is 15,000 lb. At the end flange, this force is resisted by the weld between the vacuum vessel tube and the end flange. As shown on Fermilab drawing 5520-ME-390003, this weld is specified to be a 6 mm (0.24 inch) fillet. The stress on the weld is given by

$$s_w = \frac{f_a}{(l)(t_w)}$$

where: σ_w = stress in the weld

f_a = axial force = 15,000 lb

l = linear length of weld = 113 inches

t_w = weld equivalent thickness = 6 mm/√2 = 4.24 mm = 0.17 inch

For this case, the weld stress, σ_w, is 780 psi which is well below that allowed by UW-18 of the Code given by:

$$(20{,}000 \text{ psi})(0.8)(0.55) = 8{,}800 \text{ psi}$$

4.4.3 Access ports

The access ports in the connecting rings are openings in the vessel. Section UG-37 of the Code requires that the minimum area of reinforcement for these openings is:

$$A_r = dt_rF + 2t_nt_rF(1 - f_{r1})$$

where: A_r = area required

d = inside diameter of opening = 3 inches

t_r = minimum required thickness of the shell at the design pressure computed using UG-27(c)(1) = 0.048 (see table 4.4.3)

F = correction factor = 1

t_n = nozzle wall thickness = 12 mm = 0.47 inches

f_{r1} = strength reduction factor = 1

For this case, $A_r = 0.14\ in^2$. The area for reinforcement available in the shell is given by the larger of:

$$A_1 = d(E_1t - Ft_r) - 2t_n(E_1t - Ft_r)(1 - f_{r1})$$

or

$$A_1 = 2(t + t_n)(E_1t - Ft_r)\text{-}2t_n(E_1t - Ft_r)(1 - f_{r1})$$

where: E_1 = weld efficiency = 1

t = vessel wall thickness = 1.5 inches

For this case, $A_1 = 5.72\ in^2$ from the two expressions above. Since the available area, A_1 is greater than the required are A_r, no additional reinforcement is necessary.

The access ports are attached to the vacuum vessel using fillet welds. These welds support the structural weight of the internal magnet assembly and all other internal components.

The weld supports this weight in shear. The maximum force which can be supported by these welds is given by:

$$F = \sigma_w \pi d t_w E$$

where: F = maximum allowed force in the weld

σ_w = stress in the weld = (20,000 psi)(0.8) = 16,000 psi

d = effective diameter of the weld = 3 inches

t_w = weld equivalent thickness = 5 mm/√2 = 3.54 mm = 0.14 inch

E = weld efficiency = 49% (per UW-15)

For this case, F = 10,344 lb. The largest load in any LHC cold mass is 23,500 lb shared by 8 of these welds. In that case each weld supports 2,938 lb so the weld is sufficient.

4.5 Summary

The LHC prototype vacuum vessel satisfies all requirements of the ASME Code and the Fermilab ES&H manual with the exception of the joint between the end rings and the vessel shell. The code requires this lap joint to have minimum overlap of four times the vessel thickness or 2 inches. The design overlap is 1 inch. As shown in 4.4.2 stress in the weld between the end ring and vessel shell is less than 10% of the allowable stress.

Chapter 5

LHC Interaction Region Quadrupole

Interconnect

5.0 Introduction

The interconnect is the region between the magnet and the feedbox when installed at MTF. The purpose is to transport cryogens, electrical wiring and insulating vacuum from the feedbox to the magnet. There are thirteen total bellows which make up the interconnect consisting of seven unique designs. Their descriptions and a summary of their operating parameters are shown in table 5.0.1.

Table 5.0.1. Bellows operating parameters

Parameter	**HX Outer Shell**	**Cooldown Line**	**50-70 K Shield**	**Beam Tube Differential Expansion**	**Cold Mass**	**MTF Beam Tube**	**Vacuum Vessel**
Internal Media	Lhe	Lhe	Ghe	Lhe	Lhe	Vacuum	Vacuum
External Media	Vacuum	Vacuum	Vacuum	Vacuum	Vacuum	Vacuum	Air
Operating pressure	1.3 bar	1.3 bar	19.5 bar	1.3 bar	1.3 bar	Vacuum	Vacuum
Internal Design Pressure	20.0 bar	20.0 bar	22.0 bar	20.0 bar	20.0 bar	Vac.-1 bar	Vac.-2 bar
External Design Pressure	1 bar	1 bar	1 bar	1 bar	1 bar	Vac.-1 bar	1 bar
Temperature Range	1.9 - 300 K	1.9 - 300 K	50 - 300 K	1.9 - 300 K	1.9 - 300 K	1.9 - 300 K	300 K
Minimum Cycle Life	5000 cycles	5000 cycles	5000 cycles	5000 cycles	1000 cycles	5000 Cycles	5000 cycles

5.1 Design codes and evaluation criteria

The LHC quadrupole bellows are designed according to the standards of the Expansion Joint Manufactures Association, Inc. (EJMA). All applicable requirements of the Fermilab ES&H manual as well as the ASME Code must also be satisfied.

5.2 Materials

The convolutions on all of the bellows are 316L stainless steel. All other components that make up a bellows assembly are either 304 or 316 series stainless steel.

5.3 Bellows design

The bellows fall into two categories: formed convolutions and welded convolutions. There are six formed and one welded bellows design. Five of the six formed bellows designs are similar and are discussed in section 5.3.1. The other formed bellows, the vacuum vessel bellows, has two sets of convolutions and is discussed in section 5.3.2. The cold mass bellows is the only welded bellows and is discussed in section 5.3.3. All bellows, with the exception of the vacuum vessel bellows, will be fitted with squirm protectors any time there is pressure applied to the bellows and when installed at MTF. In addition, the heat exchanger outer shell, cooldown line and 50-70 K shield bellows have integral liners to guard against failure due to elastic instability.

5.3.1 Interconnect bellows

The interconnect bellows are all similar in design, they have one set of convolutions. A typical design is shown in Figure 5.3.1.

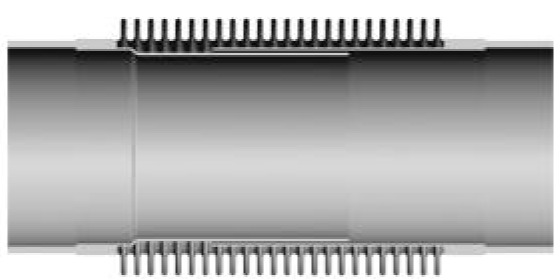

Figure 5.3.1. Typical hydroformed bellows design (cross section).

The convolutions were designed according to EJMA Section C-5.2.2. A computer program was used for the calculations. The input parameters and the results are summarized in Table 5.3.1.1.

Table 5.3.1.1. Interconnect bellows input parameters and results.

	HX Outer Shell	Cooldown Line	50-70 K Shield	Beam Tube Differential Expansion	MTF Beam Tube
Input					
Bellows ID, in.	7.00	2.25	1.88	3.00	3.00
Number of plys	2	2	2	2	1
Ply thickness, in.	0.008	0.008	0.008	0.008	0.008
No. of convolutions	64	24	24	18	18
Convolution pitch, in.	0.125	0.250	0.250	0.250	0.250
Convolution depth, in.	0.230	0.260	0.255	0.250	0.250
Design Pressure, psi	300	300	325	300	15
Travel, in	1.25	1.25	1.25	0.875	0.875
Elastic Modulus, psi	2.83E+07	2.83E+07	2.83E+07	2.83E+07	2.83E+07
Results					
Calc. stress, S2, psi	16239.9	9431.5	8892.5	12393	1239.3
Calc. stress, S3, psi	2191.3	2573.5	2758.9	2439	243.9
Calc. stress, S4, psi	52520.8	52065.9	52985.5	48522.6	4852.3
Fatigue cycles	36789.7	7743.9	6807.6	7126.3	17683.9
Axial spring rate, lb/in.	306.4	180.4	167.4	339.2	169.6
Squirm pressure, psi	36.1	28.3	26.3	71.1	35.5

EJMA requires that S2, S3 and 0.35(S4) be less than the allowable material stress which is 20,000 psi. This requirement is satisfied.

5.3.2 Vacuum vessel bellows

The vacuum vessel bellows consists of two sets of convolutions with a straight section in-between. The convolutions were designed according to EJMA Section C-5.2.2. A computer program was used for the calculations. The input parameters and the results are shown in Table 5.3.2.1.

Table 5.3.2.1. Vacuum Vessel bellows input parameters and results.

	Vacuum Vessel Bellows
Input	
Bellows ID, in.	40.25
Number of plys	1
Ply thickness, in.	0.018
No. of convolutions	8
Convolution pitch, in.	0.500
Convolution depth, in.	1.000
Design Pressure, psi	30
Travel, in	0.7
Elastic Modulus, psi	2.83E+07
Results	
Calc. stress, S2, psi	7619.6
Calc. stress, S3, psi	843.6
Calc. stress, S4, psi	39638.0
Fatigue cycles	1421083.0
Axial spring rate, lb/in.	1006.1
Squirm pressure, psi	237.0

All of the EJMA requirements are satisfied.

The straight section between convolutions is fabricated from 304 stainless steel and is considered a tube or shell under external pressure. Tables 5.3.2.2 and 5.3.2.3 below summarize the Code calculations for the straight section as an externally pressurized vessel with 1 atmosphere external pressure and as a pressure vessel with 2 atmospheres internal pressure. The MTF feedbox, to which this bellows is attached, has a pressure relief which opens just above atmospheric pressure. For the sake of the straight section acting as a pressure vessel however (table 5.3.2.3) the design pressure is defined to be 2 atmospheres.

Table 5.3.2.2 Shell as a vacuum vessel (governing equations (UG-28(c))

$$P_a = \frac{4B}{3(D_o/t)}(method1)\ or\ P_a = \frac{2AE}{3(D_o/t)}(method2)$$

Variable	Value	Units	Descriptions and References
Do	40.25	in	Vacuum vessel bellows OD
L	16.00	in	Length of shell
t	0.125	in	Vacuum vessel bellows straight section thickness
E	2.83E+07	psi	Young's modulus
L/Do	0.4		
Do/t	322		
A	0.0006		Figure 5-UGO-28.0, Appendix 5
B	7250		Figure 5-UHA-28.1, Appendix 5
Pa (method 1)	30.02	psi	Calculated maximum allowable external working
Pa (method 2)	35.16	psi	Calculated maximum allowable external working

Table 5.3.2.3 Shell as a pressure vessel (governing equations (UG-27(c))

$$t = \frac{PR}{SE - 0.6P}(circumferential stress)\ or\ t = \frac{PR}{2SE + 0.4P}(longitudinal stress)$$

Variable	Value	Units	Descriptions and References
P	30	psi	Internal design pressure
R	20.00	in	Shell inside radius
S	15,040	psi	Section VIII, Division 1, Table UHA-23, derated to 80% of allowed
E	0.60		Weld joint efficiency (Table UW-12)
t(c)	0.067	in	Minimum shell thickness when sized for circumferential stress
t(l)	0.033	in	Minimum shell thickness when sized for longitudinal stress
t	0.067	in	Minimum shell thickness

From table 5.3.2.2, the maximum allowable external working pressure of the vacuum vessel, Pa, is 30.02 psi. The minimum pressure required by the Fermilab ES&H manual, chapter 5033 is 2 atmospheres or 30 psi so the requirement is met. From table 5.3.2.3, the minimum shell thickness for the vacuum vessel bellows straight section is 0.067 inch. The straight section wall is 0.125 inch so the requirement is met.

5.3.3 Cold mass bellows

The cold mass bellows is a welded bellows. Since EJMA only covers convoluted bellows, this bellows is vendor designed per our specifications. These specifications are the operating parameters listed in Table 5.0.1. There is a bellows protector that fits on the OD of the convolutions to protect the bellows from squirm. This can be seen in Figure 5.3.3.1.

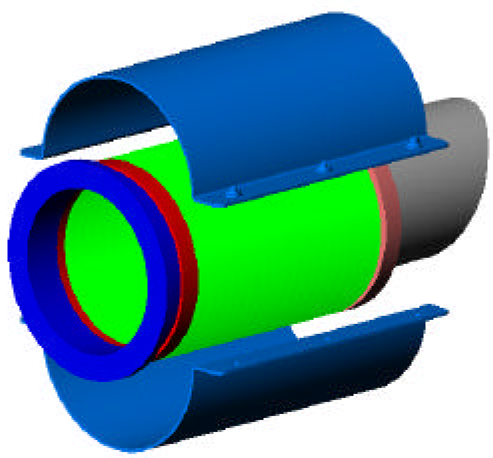

Figure 5.3.3.1. Cold mass bellows protector (exploded view).

5.4 Summary

All of the bellows to be used in the interconnect at MTF for the LHC quadrupoles meet the requirements of EJMA as well as all applicable ASME Codes and the Fermilab ES&H manual.